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November 1993
NSRP 0408

THE NATIONAL SHIPBUILDING RESEARCH PROGRAM

1993 Ship Production Symposium

Paper No. 12: A Conceptual Design Study of the Construction of Hydrodynamic Control Surfaces

U.S. DEPARTMENT OF THE NAVY
CARDEROCK DIVISION,
NAVAL SURFACE WARFARE CENTER

Report Documentation Page

Form Approved
OMB No. 0704-0188

1. REPORT DATE	2. REPORT TYPE	3. DATES COVERED
NOV 1993	**N/A**	**-**

4. TITLE AND SUBTITLE	
The National Shipbuilding Research Program, 1993 Ship Production Symposium Paper No. 12: A Conceptual Design Study of the Construction of Hydrodynamic Control Surfaces	5a. CONTRACT NUMBER
	5b. GRANT NUMBER
	5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S)	5d. PROJECT NUMBER
	5e. TASK NUMBER
	5f. WORK UNIT NUMBER
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) **Naval Surface Warfare Center CD Code 2230-Design Integration Tools Bldg 192, Room 128 9500 MacArthur Blvd, Bethesda, MD 20817-5700**	8. PERFORMING ORGANIZATION REPORT NUMBER
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)	10. SPONSOR/MONITOR'S ACRONYM(S)
	11. SPONSOR/MONITOR'S REPORT NUMBER(S)

12. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release, distribution unlimited

13. SUPPLEMENTARY NOTES

14. ABSTRACT

15. SUBJECT TERMS

16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
a. REPORT	b. ABSTRACT	c. THIS PAGE			
unclassified	**unclassified**	**unclassified**	**SAR**	**24**	

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1993

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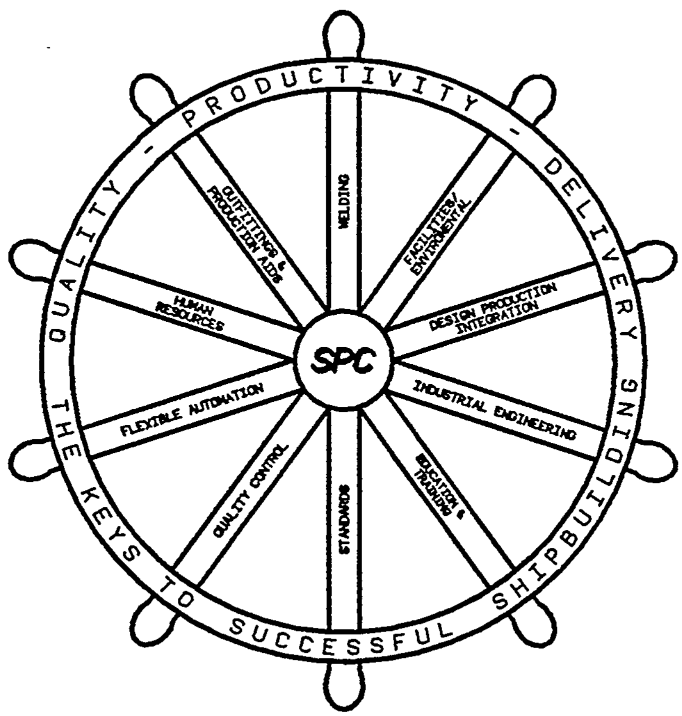

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Williamsburg Virginia, November 1-4, 1993

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A Conceptual Design Study of the Construction of Hydrodynamic Control Surfaces

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ABSTRACT

Hydrodynamic control surfaces are traditionally built as steel fabrications. While this gives a very strong structure, it is rather heavy and costly, it is difficult to achieve smooth surfaces, and the steel is susceptible to erosion, corrosion and marine fouling. This paper describes a conceptual design study aimed at creating a competitive advantage for the manufacturers of control surfaces by using modern materials in a composite structure. The conceptual design process, as applied here, starts by specifying the design requirements for the construction of control surfaces, and listing a set of criteria against which the concept designs can be evaluated. A total of six concept designs are described and evaluated in comparison with a traditional steel fabrication, and one concept is selected for further development. This comprises a light steel frame structure, with thin steel inner face plates enclosing an inner core that is filled with polyurethane foam. The surface shape is also formed with polyurethane foam poured between the faceplates and a surface mold plate. Finally, the surface is sprayed with a polyurethane elastomer coating.

NOMENCLATURE

A = area of control surface
B = balance
c = mean chord
Cp = pressure coefficient
Cr = root chord
C1 = tip chord
Ef = elastic modulus of foam
E s = elastic modulus of solid polymer
g = acceleration due to gravity
h = head of water
M = mass
p = local pressure
pd = dynamic pressure
p_a = atmospheric pressure
S = span
Tr = root thickness
Tt = tip thickness
u = local velocity
u = free stream velocity
E = sweep or rake angle
p = water density
pf = density of foam
ps = density of solid @ymer
φ = volume fraction of foam

CONVERSION OF UNITS

1 meter= 3.281 feet
1 millimeter= 0.04 inch
1 kilogram= **2.2** pound
1 Newton= 0.225 pound force
1 kilonewton = 0.1004 tons force
1 kilonewton meter =738 pound force foot
1 kilogramlcu. meter = 0.0624 lb/cu. ft
1 Megapascal = 145 psi
£1= $1.57

INTRODUCTION

Hydrodynamic control surfaces are used on ships and submarines to control ship motions, and are found in the form of rudders, stabilizer fins and hydroplanes. They are traditionally built as steel fabrications, with wood, reinforced plastics, and cast nylon used as alternatives for small size control surfaces. Some recent designs of hydroplanes and rudders for submarines

have utilized syntactic foams and non-metallic composites (l).

Steel fabrication of hydrodynamic control surfaces is well-suited to the manufacturing facilities of marine engineering companies and shipyards, and provides a control surface that can be easily repaired throughout the world using the skills and facilities of any well equipped ship repair yard. However, there are also a number of disadvantages with a steel fabrication. It is necessary to use relatively thick steel plate with internal stiffeners to achieve an accurate and smooth surface profile. This leads to a heavy construction that requires stitch welding and a fair amount of dressing of welds, all of which leads to high cost. The steel surface is also affected by corrosion, marine growth, and cavitation erosion. It therefore requires good anti-fouling, with periodic maintenance to retain the surface in good condition.

The objective is to improve the competitive position of marine engineering companies in developed nations through the use of modern technologies. The goals are to reduce the cost and weight of hydrodynamic control surfaces, while also improving the geometrical accuracy and smoothness of the surfaces, and their resistance to corrosion, erosion, and marine fouling. This has been achieved by developing a methodology for the design and manufacture of hydrodynamic control surfaces utilizing modern material and construction technologies, using a design approach to look at how best to utilize existing modern materials in a traditional product.

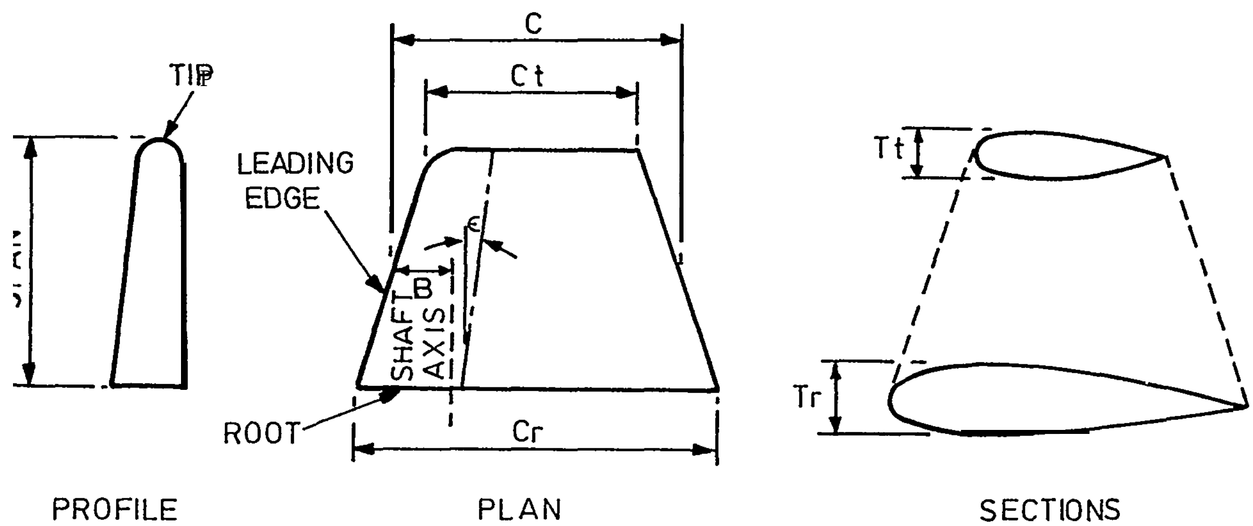
This paper reports on the conceptual design phase of the work during which many alternative design solutions were devised and evaluated, a preferred solution adopted, and then developed to the point where the detailed design of a prototype stabilizer fin could be undertaken. The bases for this work were the design requirements for hydrodynamic control surfaces, and the criteria against which the alternative concept designs could be evaluated.

DESIGN REQUIREMENTS

The design requirements for the construction of hydrodynamic control surfaces cover rudders and roll stabilizer fins for ships, and hydroplanes and rudders for submarine applications. These latter items require special consideration because of the high hydrostatic pressure loadings. The requirements have been formulated as a result of correspondence and meetings with leading UK shipyards and marine equipment manufacture about the use and fabrication of current designs, and by analysis of the loadings.

Types and Sizes of Control Surfaces.

This study is restricted to cantilevered control surfaces carried on a socketed shaft, such as ase used



= Span
t = Tip Chord
r = Root Chord
lean Chord, C = 0.5*(Ct + Cr)
alance = B/C

Tt = Tip Thickness
Tr = Root Thickness
ϵ = Rake or Sweep Angle
Aspect Ratio, a = S/C
Thickness Ratio = T/C

Figure 1. Parameters for a Trapezoidal Control Surface

for spade rudders and stabilizer fins. Other forms of hydrodynamic control surfaces, such as Mariner rudders, have less highly stressed interfaces with the ship, and it is therefore anticipated that methods of construction developed for cantilevered surfaces will be easily adapted to surfaces that are more uniformly supported. A trapezoidal plan form shape is used with the geometry of the control surface defined by the parameters given in Figure 1.

The range of sizes of control surfaces depends on the application. For commercial and naval applications (i.e. not including yachts), the following ranges represent current practice.

Trapezoidal stabilizer fins:
1.0 m to 15 m^2 (10 ft^2 to 160 ft)

Rectangular flapped fins:
2.5 m to 20 m (25 ft^2 to 215 ft^2)

Submarine rudders & hydroplanes:
10 m to 30 m (105!ft^2 to 320 ft^2)

Spade rudders:
up to about 25 m^2 (270 ft^2)

Larger rudders are usually supported on a horn.

The majority of hydrodynamic control surfaces are trapezoidal in plan form, with a raked leading edge. The shaft is fitted on an axis passing close to the center of pressure loading on the surface, so as to balance moments due to the loads. On trapezoidal surfaces a balance position behveen 20% and 30% of the chord is used, while for a flapped surface a balance of 30% to 35% is required. Aspect ratios may vary from about 0.5 to 2.5. The hydrodynamic section most commonly employed is the symmetrical NACA (National Advisory Committee on Aeronautics) 4 digit section with thickness between 12% and 33%. The thickness is determined primarily by the need to accommodate the shaft. There is slightly higher drag with thicker sections, but with the advantage of a flatter pressure distribution, giving less face cavitation. The tip shape can be faired or square.

Loadings

The following sources of loading need to be considered when undertaking the structural design of a hydrodynamic control surface.

1. Hydrodynamic loads due to the flow of water over the surface which vary with the speed and frequency of rotation of the surface about its shaft.
2. Hydrostatic loads due to the pressure head of water above the control surface.
3. Impact loads due to collision with debris in the water, or with quay struchues. The surfaces should also resist damage due to loose items being dropped onto them during manufacture, or in dry dock.

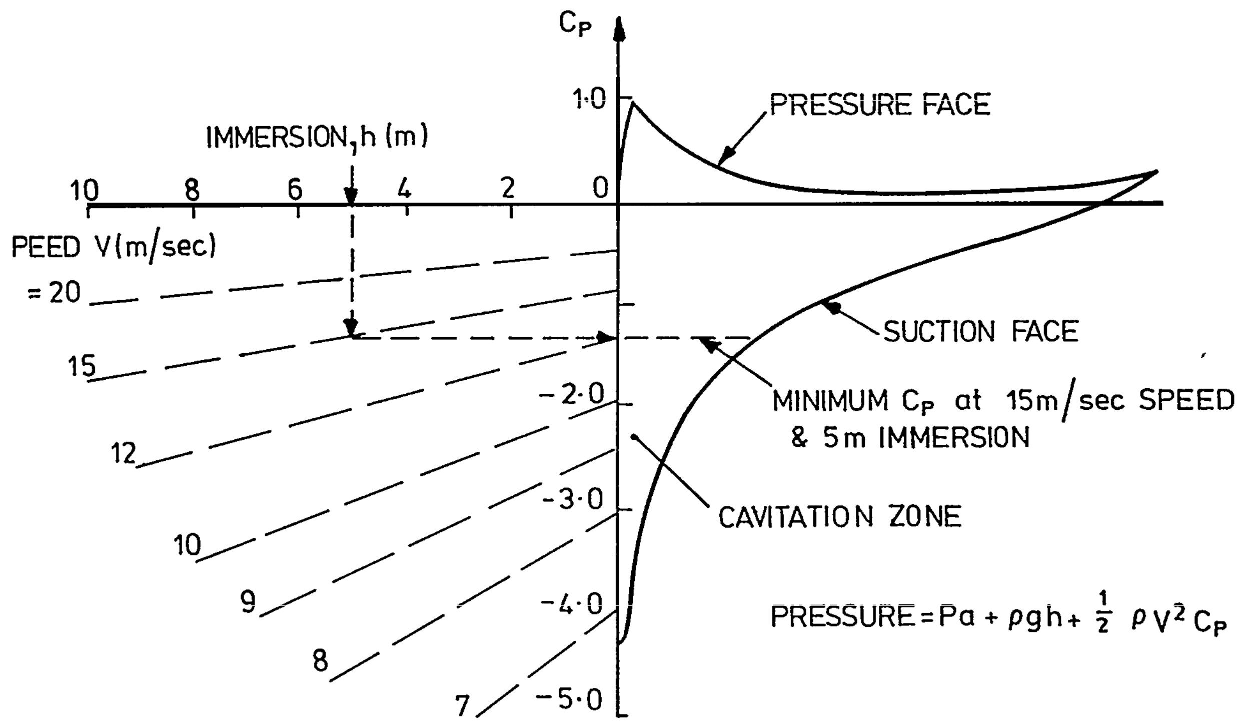


Figure 2. Pressure Coefficients on a Control Surface at Cl=1

4. Handling loads during manufacture.

5. Shock Loads. Control surfaces built for military applications are required to withstand a specified level of shock from underwater explosions.

Hydrodynamic Loading. Hydrodynamic loading arises from the pressure distribution that occurs on the control surface as a result of the variation in the velocity of flow over the surface. Figure 2 shows a typical pressure distribution for a control surface with an aspect ratio of 1.0, and an angle of incidence of 25 degrees where the pressure coefficient is given by:

$$c_p = \frac{p - p_o}{\frac{1}{2}\rho U^2} \qquad (1)$$

This figure is used for design purposes, and is based on data for NACA four digit sections given by Abbott & Doenhoff (2). The minimum value of C_p that can be sustained without cavitation depends on the immersion of the lifting surface, and is given for a range of speeds from 7 m/s up to 20 m/s (13.6 - 38.9 knots). The minimum value of C_p is applied over that part of the surface where potential values less than the minimum are indicated.

The criteria stated so far indicate worst loading conditions that are treated as static loads for design purposes, since they will usually only occur under casualty conditions (i.e. when a fault causes maximum angle of incidence at maximum ship speed). Fatigue loads will be lower than the maximum static loads, since it is normal practice to limit angles so as to prevent C_p from reaching the cavitation limit under normal working conditions. Design of the structure of a hydrodynamic control surface for a specific application should, of course, be checked against the loads that will be experienced in that application.

Hydrodynamic Loading Limits. The pressure loading that a particular hydrodynamic control surface can carry is limited by the ability of the cantilever shaft to carry the bending moment induced by the pressure loading, and the control surface must be operated so as not to exceed this loading. By assuming an allowable fatigue stress of 150 MPa (22,000 psi), which is typical current design practice, approximate maximum working loads are derived and shown in Figure 3 for a control surface with any combination of area, aspect ratio, and section thickness ratio. The casualty load is taken as approximately twice the working load, although it maybe higher in high speed ships.

Torsional Loads. The torsional load on a hydro- dynamic control surface is given by the first moment of the pressure loading measured about the shaft axis. The maximum pressure load on a control surface is only slightly influenced by the rate at which the angle of incidence is changing (called the slew rate), but the center of pressure is considerably influenced by the slew rate, which must therefore be taken into account in the torsional load calculation. A slew rate which acts to increase the angle of incidence causes a small increase in the pressure loading, and a movement of the center of pressure toward the trailing edge of the control surface. Due to a paucity of data on this effect, empirical estimates of torsional load are commonly used, with the relevant parameters being area, chord and slew rate of the control surface, and ship speed. The balance is chosen to minimize torsional load. With the load as derived from Figure 3 the maximum working torque is given approximately by

t= FC(0.03+0.01 C(s over v)) (2)

where t = torque in kN.m (lbs.ft)
F= load in kN (lbs)
c = chord in m (ft)
s = slew rate in degrees per second
v = ship speed in m/s (ft/sec)

For casualty conditions the load should be doubled and the factor 0.03 increased to 0.045.

Impact Loading. The requirement to resist impact load in service is specified in terms of a collision at design speed with a tree trunk 120 mm

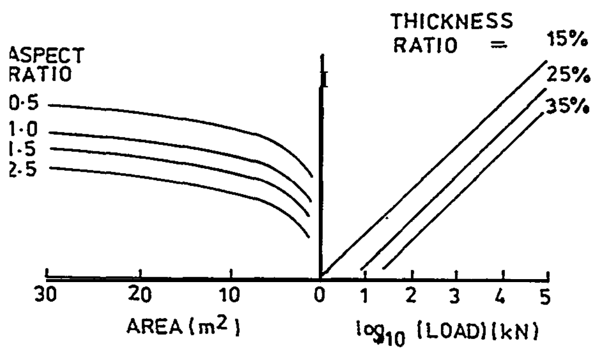


Figure 3. Hydrodynamic Load Limits

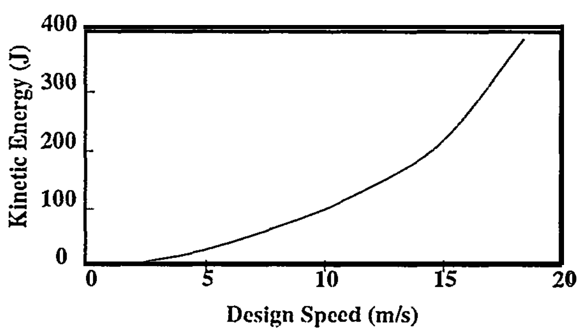


Figure 4. Design Values for Impact Loading

(4.7 in) in diameter and 2 m (6.6 ft) long. The effects of the impact load depend on the contact area between the object and the surface, and on the overall stiffness of the control surface. Estimated values of the impact loads at different design speeds are shown in Figure 4.

For submarines, serious collisions can occur with quays when docking. This is because the surfaces extend beyond the beam of the submarine, so the probability of impact damage increases for submarine fins, hydroplanes and rudders. Special attention to impact loads should therefore be included in the evaluation of submarine control surfaces.

Requirements for loadings due to grounding are not specified, and will not be considered in the design criteria.

Fatigue. The design life for stabilizer fins is 20 years, which is assumed to correspond to 10^7 cycles at maximum working load with a frequency of 0.1 Hz. For submarines the plamed life is 25 to 30 years, but a duty of 10^7 cycles is also specified due to a lower utilization rate. Rudders are only occasionally used at maximum loading.

Factors of Safety. For metals, the factor used for service load is 3.125 on yield strength, or 5.0 on ultimate strength, while for casualty loads, a factor of 1.5 on yield strength is used. For fiber reinforced plastic, factors from 4.0 for static loads to 6.0 for fatigue loads are used as recommended by Chalmers (3).

General Construction Requirements

Surface Coatings. These should be at least as resistant as the painted steel surface that is currently used. The coating should remain in good condition after 3 years in service, and should provide sufficient protection to ensure that the surface structure will still be repairable after 6 years without any intermediate maintenance. Surface coatings should be resistant to cavitation erosion and marine fouling.

Accuracy of Profiie. Geometric tolerances should be such that all parts of the lifting surface will be within ± 0.003 sq.root of A (A= fin area) of the drawing size.

Surface Smoothness. Surface smoothness can be measured by holding a flexible batten (approx. 500 mm (20 in) long) against the surface and checking that a designated feeler gauge cannot be inserted between batten and the surface: for the forward 25% of the surface, the gauge should be 2.5 mm (0.1 in) thick, for the remainder of the surface, the gauge should be 5.0 mm (0.2 in) thick. Additionally, the rate of deviation of the surface of the plate from a smooth curve should not exceed a slope of 1:10 on the forward 25% of the surface and 1:5 on remainder of the surface.

Shaft Requirement. A removable shaft is not an absolute requirement even though they are fitted to many control surfaces for ease of repair. While there is no design constraint on the shaft interface, the shaft must be perpendicular to the root plate and fitted at a balance of between 20% and 35%. The shaft must be circular at the hull line.

CONCEPTUAL DESIGN APPROACH

The approach adopted for the conceptual design process involves the following steps.

1. Derive a set of criteria from the design requirements for use in evaluating the relative merits of the various concept design proposals. It is important that the criteria should be set prior to devising the concept designs, so as to avoid subconsciously writing the criteria in terms advantageous to a favored concept.
2. Devise a set of values for the design parameters to define a standard control surface based on a typical application for use in comparative evaluation of the concept proposals. The application chosen for this study is fin stabilization of ship roll.
3. Divide the problem into a number of sub-problems based on functional elements of the structure of the control surface.
4. Devise a number of solutions to each sub-problem, and evaluate their feasibility and cost.
5. Use various combinations of the sub-problem solutions to create a number of concept design proposals. Test for feasibility and discard any that are not viable.
6. Generate data for each concept design proposal for the standard control surface defined in step 2.
7. Evaluate the alternative concept design proposals against the criteria defined in step 1.
8. Select the best design solution,
9. Develop the selected design.

Evaluation Criteria

The evaluation criteria can be grouped into three categories relating to technical, manufacturing and commercial factors. Criteria within each of these categories are listed below with a discussion of desirable characteristics.

Technical Criteria

Mass of the surface. A specific objective of the project is to reduce the mass of control surfaces by a target of 25% relative to current fabrication

methods. The mass of each concept is calculated and indicated in the evaluation chart The mass of a steel fabricated trapezoidal fin is given approximately ax

$$M = 350 AI^{fi} \text{ kg } (= 40\ Al''^{x} \text{ lbs}) \qquad (3)$$

where A is control surface area in meters^2(feet2).

Accuracy of Surface. Concept designs are ranked according to the ease of achieving the design requirement for accuracy.

Appearance of Surface. Users expect control surfaces to be smooth, and this can also be important for the hydrodynamic performance, although the NACA 4-digit sections are quite tolerant to surface roughness.

Resistance to Erosion (Cavitation). Some degree of cavitation will nearly always occur on a control surface (unless deeply submerged), even if only within the tip vortex. Painted mild steel is not good at resisting erosion and some improvement is desirable.

Resistance to Marine Foaling. Skin materials must either resist marine fouling, or be compatible with marine anti-fouling paints. The latter is desirable in any event since all surfaces are liable to be painted, whether or not intended.

Resistance of Skin to Impact. The skin should not be breached by impact with underwater, or floating objects, particularly if the substrate material may be damaged by exposure to sea water.

Resistance of Materials to Sea Water. Materials should not deteriorate on contact with sea water, nor should they absorb more than a minimal amount of water.

General Resistance to Impact. Control surfaces should have the greatest possible resistance to impact loads without damage to the main shaft bearings and actuation equipment and withoutbreaching the water-tight integrity of the ship. The sequence of damage under increasing impact loads should be

1. impact energy absorbed without damage
2. damage to skin and immediate substrate without urgent need for repair
3. collapse of control surface structure
4. bending of shaft
5. damage to bearings
6. failure of seals, or structural damage, leading to flooding of the ship.

Resistance to Hydrostatic Loads. An estimate is given of the maximum depth in meters at which the control surface can safely operate.

Other Technical Criteria. The concept designs are each ranked according to resistance to static loads; resistance to fatigue loads; resistance to shock loads; life; shaft joint integrity; overall design integrity.

Manufacturing Criteria

It is necessary to evaluate the technical risk involved in the use of the materials proposed for each concept design, and also the extent to which the use of the materials will either require new methods to be introduced by the manufacturer, or involve subcontracting of all or part of the process.

Materials. The concept designs are ranked according to their current use in marine engineering, their use in similar applications and on a similar scale, or if they represent a new development.

Manufacturing Method. The concept designs are ranked in order of preference for:

1. methods currently used in marine engineering
2. methods that could be introduced with low training and facilities cost
3. methods that could be introduced with an investment in new staff, training and facilities
4. work that would need to be subcontracted.

Commercial Criteria

Estimates are made for each concept design of the total manufacturing cost and the cost of the control surface alone, the objective being to reduce the cost of the control surface by 40% relative to the steel fabrication; the direct material cost, excluding the shaft; the cost of all subcontract activity, including any transport or other costs associated with the subcontract; and the direct manufacturing hours, excluding those for the finshaft. These are included in the total costs at a rate of £35 ($55) per hour for machining activities and £25 ($40) per hour for fabrication activities.

STANDARD FIN PARAMETERS

As part of the overall project a prototype stabilizer fin has been built with the same geometry as an existing design fitted to a British fisheries protection vessel. This is a trapezoidal fin area of 1.5 m^2(16 ft^2) area, with the following geometric characteristics:

Fin sections:	NACAO015
Aspect ratio:	1.0
Taper ratio:	0.488
Fin shaft balance:	26.5%
Zero rake chord:	32.2%

Fin Area (m^2)	1.5	5	*10*	*15*	(X *10.76fi?)*
Span (mm)	1225	2250	3160	3870	(*x* 0.04 in)
Mean Chord (mm)	1225	2250	3160	3870	(x 0.04 in)
Root chord (mm)	1646	3024	4247	5201	(x 0.04 in)
Maximum fin angle (deg)	25	25	25	25	
Shaft Diameter (mm)	170	260	390	620	(x 0.04 in)
Normal Load (kN)	66	220	480	1300	(X 225 lbf)
Hydrodynamic torque (kNm)	7.5	57	190	820	(X 720 Ibf.ft)
Casualty load (kN)	186	620	1350	3660	(X 220 lbf)

Table I. Fin Parameters

Leading edge rake: 12.5°
Trailing edge rake: 25°

The conceptual design evaluation is carried out primarily on the 1.5 m^2 (16 ft^2) fin, with results also being extrapolated to a range of fin sizes up to 25 m^2 (270 ft^2). The parameters for the full range of fin sizes considered are given in Table I.

FUNCTION ANALYSIS

For the purposes of the conceptual design the overall problem is divided into four sub-problems by examining the primary functional requirements of a control surface. These require the provision of: a rigid hydrofoil surface, a structure to carry the hydrodynamic forces from the surface to a shaft interface, a shaft to transmit the forces back into the ship's hull structure, and an interface between the control surface and the shaft. A number of solutions to each sub-problem are devised and evaluated prior to using them in various combinations to produce overall concept design proposals. The requirements for each function are given in the following sections together with a list of the alternative solutions considered.

Rigid Hydrofoil Surface

This functional requirement to retain its hydrofoil shape under load, must also provide a smooth surface of good appearance which will resist corrosion, erosion and marine fouling. Solutions considered include:

1. rolled steel or aluminium plates, welded and stiffened, dressed smooth, primed and finished with anti-fouling
2. composite foam core sandwich structure with corrosion resistant metal, or reinforced plastic face plates
3. glass reinforced plastic layup onto a male former
4. resin and filler skin formed between a male former and a mold
5. foam outer body, formed between an inner skin and a surface mold, finished with sprayed polyurethane elastomer coating
6. outer body built up onto an inner skin using a filler material, and spray coated with polyurethane elastomer.

Hydrofoil Structure

This functional requirement must provide support to the hydrofoil surface to transmit the forces, due to the hydrodynamic loading on the surface, back to the shaft interface. Solutions include:

1. steel or aluminium webs, frames, tubes and sections
2. a solid foam or plastic core material
3. a steel or non-metallic frame to support the perimeter of the surface structure.

Shaft

This functional requirement must allow transmission of loads from the control surface into the ship via a sealed hull opening. Solutions include:

1. a tapered or straight high tensile steel shaft with circular sections throughout
2. a short circular steel shaft bonded to a filament-wound reinforced plastic tube inside the control surface
3. a continuous filament wound reinforced plastic tube bonded to an outer sleeve to bear against the seals and bearings within the ship
4. a steel shaft with circular sections in way of the seals and bearings, and square sections in way of the shaft interface
5. a steel shaft with circular sections, welded to a steel box at its outer end.

Shaft Interface

This functional requirement must provide for a rigid connection capable of transferring the torsional, direct and bending forces from the hydrofoil structure to the shaft. Solutions considered include

1. a cast steel socket taper or parallel bored to provide an interference fit with the shaft, which is retained in position by an end nut, or else shrink fitted
2. a square section tapered steel torsion box running the spanwise length of the control surface, and adhesively bonded or welded to square end sections of the shaft, and also retained axially by an end bolt
3. a parallel or tapered circular hole bored into the solid core of the hydrofoil structure, into which the shaft is adhesively bonded.

Concept Proposals

Six new concept design proposals are devised horn viable combinations of solutions to the four sub-problems described above. These, together with the traditional steel fabrication, are described and evaluated in the following sections, using the design of the prototype roll stabilizer fin as a basis for the evaluation.

CONCEPT NO. 1- STEEL

Basic Form of Construction

As illustrated in Figure 5, webs, frames, root plate and tip plate are welded together, and to a shaft socket to form a steel skeleton. Steel skin plates are fillet welded on one side of the fin, and slot welded on the closing side. If required, a cast steel fairing can be welded to the tip of the fin. All surface welds are dressed to satisfy the requirements for surface smoothness. Internal surfaces are coated with an epoxy, and may also be filled by injecting polyurethane foam. The outside surface is shot blasted, primed, painted, and treated with an anti-fouling paint. The shaft is fitted to the tapered, or parallel, bore of the socket with a key, or an interference fit.

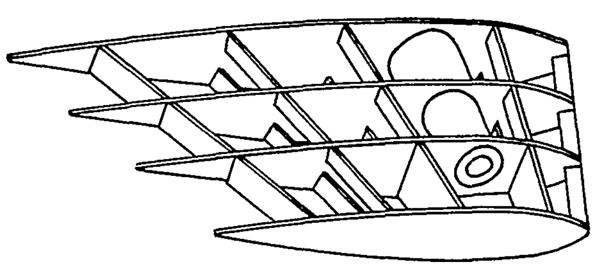


Figure 5. Fabricated Steel Structure

Alternative Approaches

Other metals can be used. For example, using aluminium will result in weight reduction, since material thicknesses are determined by stiffness requirements and corrosion resistance. However, such designs are only used on aluminium craft since the lower cost of steel is otherwise preferred.

It may also be possible to attach skin plates by bonding rather than welding. This allows thinner steel plating to be used since there will be less distortion, and also removes the requirement to dress the welds. However, because the webs and frames require flanges to provide sufficient bond area, the total cost is unlikely to be lower.

Discussion

Mild steel structures are well understood, have a proven track record of reliable service, and offer no particular fabrication hazards. Most shipyards and marine engineering workshops can undertake this type of fabrication so repair facilities are readily available. This concept represents the standard manufacturing method for control surfaces, having superseded steel castings some years ago.

Good surface accuracy can be achieved, but only at a cost of thick skin plates (to reduce weld distortion), closely spaced internal structure (typically 40 times plating thickness), and surface dressing. This leads to both high labor content and a heavy structure. Although the resulting appearance is good, it will deteriorate with erosion and fouling, and is therefore dependent on the performance of paint coatings, which are themselves quickly damaged by cavitation erosion. Surface life is therefore primarily dependent on satisfactory maintenance. Sprayed polyurethane elastomer coatings could greatly improve this situation, and reduce the need for cavitation and corrosion allowances. Both steel skin and core have good impact resistance up to the point at which overall damage owns to the shaft bearings or seal assemblies. The thick plating required for a good surface profile is more than adequate to resist fatigue and hydrostatic pressure loads, while for submarine applications, the control surface can be he flooded to balance hydrostatic loads.

Overall design integrity of the shaft/fin interface is good, provided a socketed design is used rather than a palm end. Shaft to socket joints (either keyed or interference fit) are reliable, and there are no problems in load transfer from socket to fin as a result of the integral nature of the design.

CONCEPT NO. 2- STIFF GLASS REINFORCED PLASTIC (GRP)/FOAM SANDWICH SKIN

Basic Form of Construction

The philosophy behind this concept is that the skin of the fin should be made sufficiently stiff to carry all the hydrodynamic loads without the need for any stiffeners. A simple structure can then transfer the hydrodynamic loads from the skin back to the shaft, as shown in Figures 6 and 7.

The fin/shaft interface comprises a steel box with a web at each end which is welded to the shaft well away from the high stress areas. The box thus encircles the shaft, and provides a large flat area for the bonded interface with the non-metallic structure of the fin. A non-metallic bush with a low modulus of elasticity is fitted where the shaft passes through the root plate, to provide some support without inducing high stresses. The non-metallic supporting structure for the skin is shown in Figure 6. It comprises two main webs, fitted to either side of the shaft box, which extend over the full outreach of the fin. Additional webs are fitted near the nose and tail of the fin, extending between root and tip plates which are cut slightly smaller than the finished section shape of the fin. The webs are bonded to the root and tip plates, with frames fitted between the webs to provide lateral support. A frame may also be required at approximately half span to help transmit the loads. The webs and frames are all cut from a thick (e.g. about 30 mm (1.2 in)) sheet so as to provide a sufficient bond area at their edges. The edges of the webs and frames are cut straight and lie about 40 to 50 mm (1.6-2.0 in) blow the finished surface of the fin such that, when bonded into position, they lie on three plane surhces on each side of the fin forming a land for the flat skin plates.

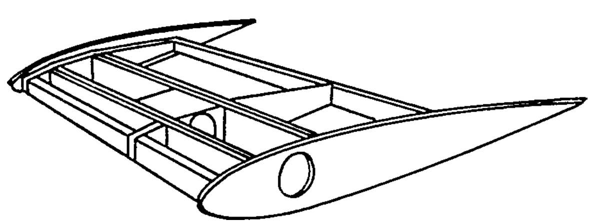


Figure 6. Frame Structure for Concept No. 2

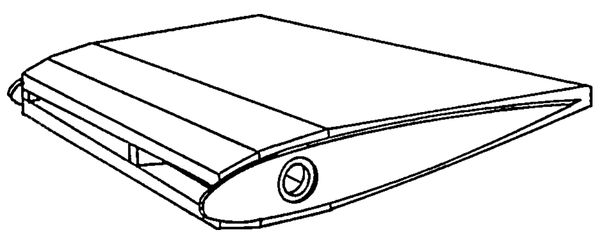


Figure 7. Skin Plates for Concept no. 2

The stiff skin is formed by laying up GRP onto one side of a suitable foam sheet from which plates are cut to fit the three planes on each side of the fin. These plates are then fitted by bonding their GRP skins to the underlying webs, and bonding their edges together. Additional foam blocks are bonded to the nose and tip of the fin. The foam on the skin plates now projects beyond the finished surface of the fin (see Figure 7), and can be cut back to the required shape by guiding a cutter between the root and tip plate sections, such that the cutter always lies along a constant chord line.

After shaping, the foam represents a male former for the fin, slightly smaller than the required size of the fin. A GRP skin is then laid up directly onto this male former.

Alternative Approaches

A sprayed application of the outer skin may be possible. This could consist of chopped glass filaments in epoxy or polyester resin, or possibly polyurethane sprayed onto a glass weave already laid onto the surface. Alternatively, resin injection molding or other vacuum forming techniques could be used with a variety of fiber/resin systems to form the skins. A variety of materials could also be used for the root and tip plates, webs and spacers. These could be cut from PVC sheet, formed from rigid foam, or fabricated as steel sections.

Discussion

An accurate surface profile can be produced without joints, which is smooth enough for practical purposes. Although further data is needed, resistance to cavitation erosion is not expected to be any better than steel, even though subsequent corrosion will not be involved. A sprayed polyurethane coating would give an improved erosion resistance. The relatively thin GRP skin would be subject to damage from localised impact forces, which may lead to water ingress and skin delamination, but minor damage should be easily repaixed. Under general impact or shock loading, the inherent strength will be less than steel, but greater flexibility and localized collapse may prevent serious damage to shaft, bearing and seal assemblies.

The lightweight GRP/foam sandwich structure will be strong enough to resist moderate hydrostatic and overall dynamic loads provided that all components are designed to resist the applied loads with an adequate factor of safety. Care must be taken to allow for ageing and fatigue resistance of the plastic components, with recommended working stress levels as low as 15% of static strength. There will be no surfeit of strength, as is present in steel structures, but a 10^7 cycle life should be achievable if water penetration is minimized through well-controlled GRP

lay-up and cure. As it is undesirable to free flood such a structure, the design will be limited by depth.

The integrity of the shaft/fin joint is dependent on minimizing stress concentrations by controlling the quality of the welded joint between the shaft and the central steel box. Overall integrity is dependent on the bonds between this steel box, the webs, and the sandwich skin. Large bond areas are possible, so mean stresses should be low, but care will be required to avoid stress concentrations at the box corners, where optimum bonded joint design may be difficult to achieve. Although this concept has been used for small scale fins on SWATH vessel (4) it cannot yet be considered proven technology at larger scales.

The materials involved include steel, plastics, foams, resins, fibers and adhesives. All are currently in marine use worldwide, but special care is needed when handling resins and adhesives. Hand lay-up of GRP is also labor intensive. Accurate cutting and shaping methods are required for the foam components, and the sandwich core may require a specially designed cutting tool. Accurate assembly will require jigs and clamping systems (e.g. vacuum bags) during adhesive cure. Despite having to cure the resins and adhesives, an overall production time of four weeks should be achievable.

CONCEPT NO. 3- INNER STRESSED SKIN

Basic Form of Construction

A strong steel torsion box runs the spanwise length of the fin and carries the main bending and torsional loads back to the shaft. As shown in Figure 8, steel root and tip plates are welded to each end of the torsion box. They are connected at the leading edge of the fin by a nose bar, and at the trailing edge by a tail bar and a tail plate to form a rigid steel structure. To complete the steel fabrication, thin steel face plates are welded to the structure to form forward and aft void spaces. The shaft, which has a square tapered end section, is socketed into the fabricated torsion box and adhesively bonded into place. The forward and aft void spaces are filled with high density (200 kg/m^3 (12.5 lb/ft^3)) free rise, closed cell, rigid polyurethane foam which acts as a structural component to transmit shear forces. Foam nose and tip blocks are bonded to the steel core structure which then has surface mold plates clamped around it, while additional polyurethane foam is poured into the cavities to take on the finished hydrofoil shape of the fin. Finally the entire surface of the fin is spray coated with polyurethane elastomer to a thickness of 3 mm (0.12 in).

This proposal combines the use of modern materials with conditional steel fabrication so as to separate the load carrying function of the steel structure from the requirement to achieve an accurate and robust surface.

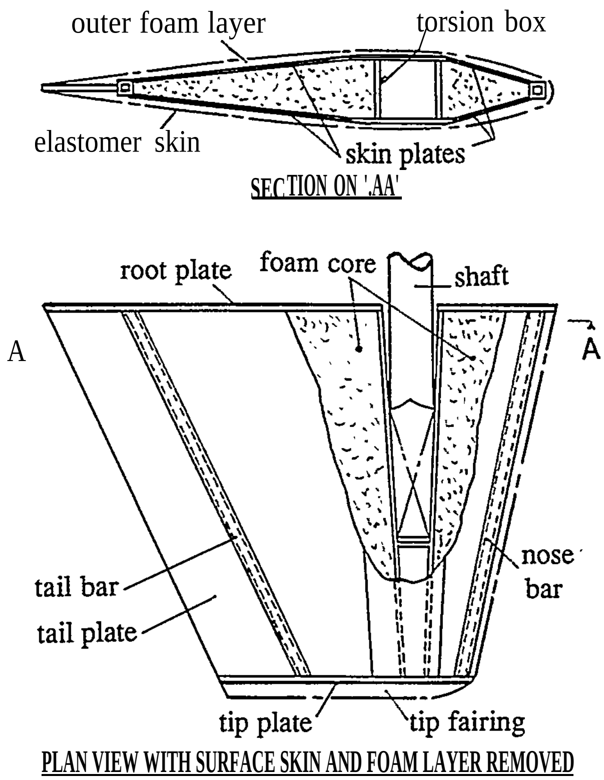


Figure 8. Arrangement for Concept No. 3

Alternative Approaches

The shaft can be welded to the torsion box. The advantage of this is that the close fabrication tolerances required for the adhesively bonded joint between the shaft and the torsion box can be relaxed. However this would be at the expense of the complexity of the structure, and would constrain the choice of materials for the shaft.

The outer layer of foam might be applied by spraying. The foam densities achieved with this process are not as high, and the final surface would be quite rough unless it was machined after foaming. Alternatively, the outer layer might be formed by trowelling on a resin based filler to build up the required surface profile, and provide a harder surface at a penalty of higher cost and mass. Both approaches avoid the cost of mold plates, but suffer similar difficulties of ensuring good accuracy.

Discussion

The labor cost of this structure is low because of the small number of welded parts, requiring the minimum of machining, with no need to dress the welds. Surface accuracy is a function of the rigidity of the mold plates used to resist the pressures created during foaming. High accuracy is possible with a rigid

mold. The polymethane elastomer can be sprayed to a high quality semi-gloss finish which does not require further protective coatings. If sprayed too quickly a mottled surface will result. The elastomer coating is highly resistant to sea water, erosion, fouling and delamination from its low modulus foam substrate, and is also unlikely to be penetrated by impacts lower than those which would cause damage to the steel substructure. The structure is therefore robust and tolerant of minor defects in the foam. However, local indentation may occur in the foam substrate iiom even relatively low impacts. In such circumstances, the old layer can be patched, shaped and the elastomer skin made good. Absorption into, and penetration of, the outer foam layers by water may well occur if the elastomer skin is damaged, even though the foam is closed cell. More critical, however, is the potential for penetration of water to the highly loaded inner steel/foam interface where deterioration of bond strength may affect structuml integrity. This is an area where further data is required.

The resilience and flexibility of the structure should attenuate shock loading well, and although the shock capability has not been calculated, it should be possible to design to a specific requirement.

The thin steel clad sandwich structure carries the stresses efficiently, and results in a lightweight fin which should be able to absorb substantial impact before damage is transmitted to the shaft or hull of the ship. Overall integrity of the shaft/fin connection, being mechanically keyed and bonded primarily in compression, is good. The integrity of the fin as a whole is largely determined by the fatigue resistance of the plate welds, the working stress in the foam being only about 0.2 MPa (29 psi). A life of 10^7 cycles is therefore a realistic design criterion. Polyurethane foams are pressure resistant to about 2 MPa (290 psi), but for higher hydrostatic pressures syntactic foams would be necessary.

Although foam and elastomer technologies are new to most shipyards, all materials are readily available, and have a proven track record in the marine environment. Shrinkage and creep of the foam materials can cause problems during manufacture which require experience to control. Specialized facilities are therefore recommended for foaming and spray coating. This process will take about a week regardless of size; one additional week is required for surface curing. The main activity is the steel fabrication which, except for the tolerance control of the torsion box, is conventional. Depending on size, this suggests an overall production time of four to six weeks.

CONCEPT NO. 4- SOLID SYNTACTIC FOAM

Basic Form of Construction

As shown in Figure 9, one or more solid large blocks of prefabricated syntactic foam with a density of about 700 kg/m^3 (44 lb/ft^3) are bonded together and then profile machined to the required fin shape. The shaft is then bonded into a bored hole, extending to the tip of the control surface in order to carry spanwise bending loads. A flexible polyurethane elastomer coating is applied to approximately 3 mm (0.12 in) thickness to form the surface skin.

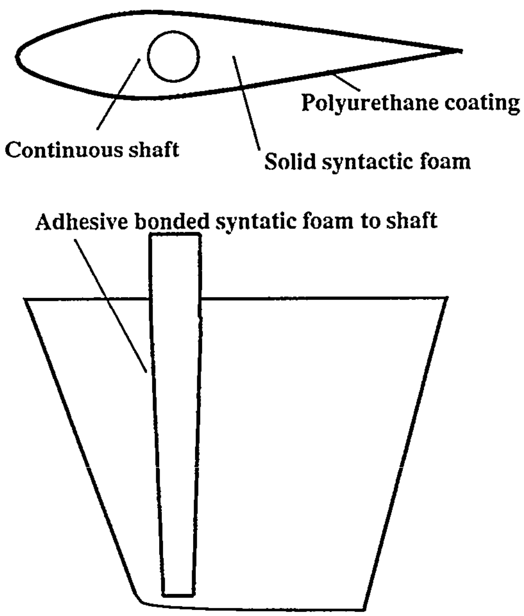


Figure 9. Arrangement for Concept No. 4

Discussion

The overall structure is relatively light, despite the need for a full span shaft, and is also simple to fabricate, with low labor costs. Syntactic foam can be easily machined using carbide tools to give an accurate surface shape but, as the polyurethane coating will reproduce the surface texture of the foam, high resolution machining is required for good surface appearance. The general performance of the elastomer coating is the same as for Concept No. 3.

There is extensive experience of syntactic foam used to resist hydrostatic loading to depths of 2000 m (6560 ft), and this concept has an estimated safety factor of four against failure under the casualty load. There is little data on fatigue performance of syntactic foams, but performance is expected to be better than fiber reinforced composites, and a 10^7 cycle fatigue life should therefore be readily achievable. Bonding tests between steel and syntactic foam adherents using cold cure epoxy adhesives indicate that joint strength is limited by the strength of the foam. The shaft to fin joint, and overall fin integrity are therefore limited by the strength of the foam, which may fracture or crush under shock or localized impact respectively.

Full repair capabilities are not yet available on a world-wide basis.

Although in regular use for high pressure buoyancy, syntactic foam is a new material for most control surface manufactures. It can either be purchased as blocks or foamed in house. Cutting and shaping present no particular problems. Polyurethane elastomers, on the other hand, are best sprayed by specialist subcontractors who can better deal with the hazards. Overall production time is three to six weeks once the foam blocks are available. Exothermic reactions during cure limit the size of blocks and speed of manufacture. Large fins must be built up by bonding layers of blocks together.

Syntactic foam has been used in French Navy designs for hydroplanes, and has also been recommended by a design study on non-metallic hydroplanes undertaken by the British Navy (l). Both these designs employ a resin or GRP skin, and require the use of a mold. However, for near surface applications, this concept is only cost effective for smaller control surfaces because of the high cost of syntactic foam.

CONCEPT NO. 5 - GRP AND FOAM

Basic Form of Construction

A prefabricated GRP filament wound tube is bonded over the end of a short shaft to form an extended shaft. This extends to the tip of the fin, and the fin is fabricated around it as shown in Figure 10.

A GRP plate, with a sealing joint around the shaft is fitted at the root. Foam blocks of low density (120 kgJm^3(7 lbs/ft?)) are bonded onto the inner support in the lower stress areas, whereas syntactic foam is applied in the higher stress areas such as the nose. Finally, the foam is shaped and GRP is laid up to form the outer skin.

Discussion

The performance of the GRP skin is as described for Concept No. 2. The general peformance will be similar to that described for Concept No. 4, but the improved load carrying capabilities of the GRP skin provides better resistance to external loads, impact and shock, although the presence of polyurethane foam will reduce the capacity to withstand hydrostatic loads. The performance of the filament wound GRP shaft, and its bonds to the steel shaft, would need to be proved by prototype testing, although a small angle machined scarf joint appears to be both practical and effective. The resulting structure is lightweight, and relies on technology fairly well understood within the marine industry. However, the large number of components requiring careful surface preparation for bonding implies some additional machining and labor costs, although resin injection systems may be able to minimize the GRP lay-up costs.

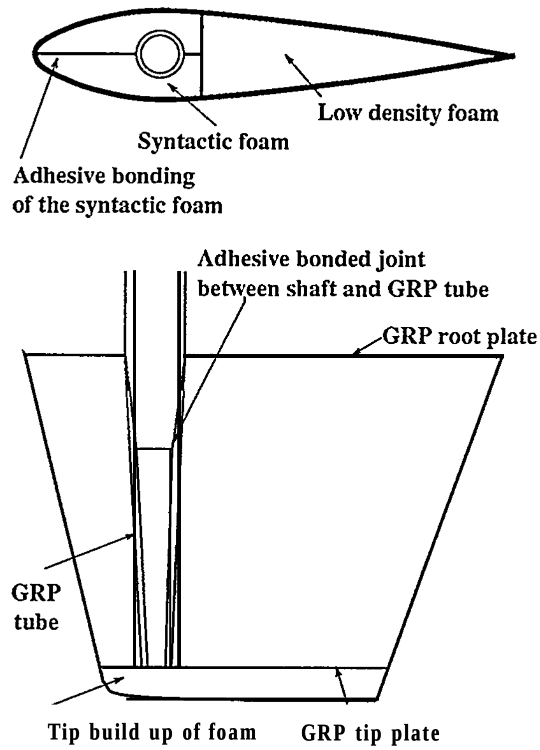


Figure 10. Arrangement for Concept No. 5

CONCEPT NO. 6- CORED OR HOLLOW CASTING

Basic Form of Construction

The fin is cast within a female mold, which provides the required finished external shape. The mold is held vertically with the tip down and the root uppermost, so that the casting material can be poured into the mold around the shaft and cores, which are suspended in place as shown in Figure 11. Various materials can be used for the casting, but for this concept description nylon 6 has been selected, as there is a significant amount of experience of its use in large castings.

The shaft is forged into a spade end to ensure that it is securely keyed into the fin, so that the torque will be effectively transmitted, and to allow a sufficient depth of material between the shaft and the surface of the fin. The cores are designed to provide uniform thickness of the cast material, and to minimize residual stress concentrations caused by the shrinkage of the cast material as it cools.

Although a mold is required, a master mold could be used to produce a range of fin sizes to a standard profile by inserting a tip mold as a lower dam, and then pouring the mix to the required depth. To reduce stress concentrations, the cores should collapse easily to accommodate shrinkage of the mix as it

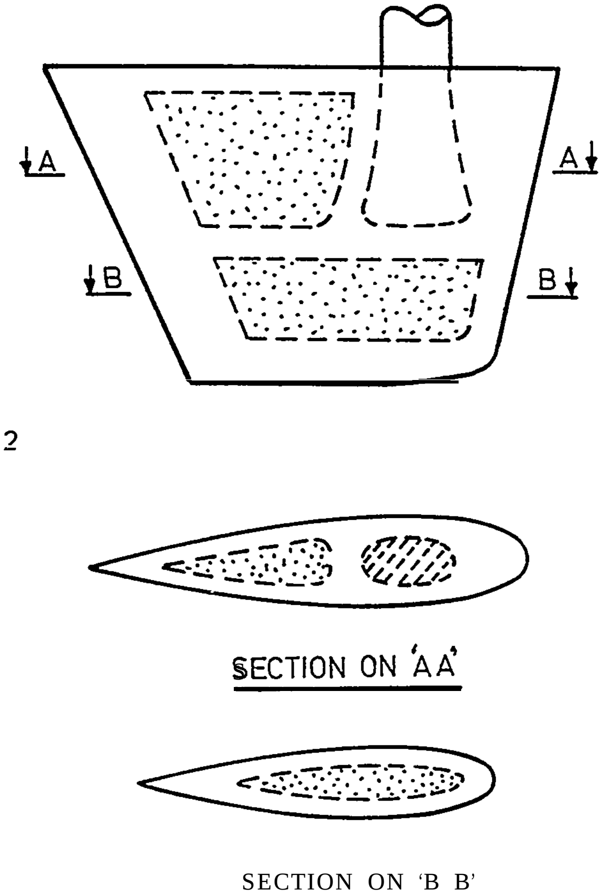


Figure 11. Arrangement for Concept No. 6

cools. It would also be an advantage if a method could be devised whereby a thin compressible coating is applied to the spade shaft, as this would allow the shrinkage to take place without inducing high residual stresses at the shaft interface. The casting should be annealed to further reduce residual stresses.

Alternative Approaches

A range of other plastic materials can be considered instead of nylon 6, and it may be possible to consider using machined flats on the shaft rather than a forged spade. At the other extreme, if weight is not a problem, metal castings can be used.

Discussion

Cast nylon offers a lightweight solution with high surface accuracy, good self-colored appearance, and low labor costs. It is widely used in the marine industry, and is used for small fin stabilizers as a solid casting (Concept 7). Although it absorbs some water, this does not appear to be detrimental. Erosion and fouling resistance are good, although not as good as polyurethane elastomer. However, general resistance to impact is less certain as there seems to be a threshold above which the material may split or shatter. Despite this, the body of the casting should withstand shock loads due to its inherent flexibility, although the behavior of the shaft/fin interface is less certain. Provided that the wall thickness of the nylon is huge, there are good margins of strength against static, hydrostatic, and fatigue loads, although the effects of ageing and fatigue require that the fin should operate at low stress levels to achieve a life of 10^7 cycles.

Overall design integrity appears to be good, particularly with a shaft interface based on a spade or flats machined on a circular shaft. Unpublished test results suggest the bond of nylon to steel is as strong as the nylon itself. The main area for concern remains the reduction of residual stresses in nylon after casting, which are a potential risk if further machining is needed as the material may shatter. This problem requires further research.

Cast nylon 6 is readily available, but generally only in casts of up to about 500 kg (1100 lb). Larger sizes would require extensive capital investment which would relegate its use to specialist subcontractors. The major advantages of this concept are its light weight and its potential low cost once the mold is made, although the concept becomes relatively expensive in large volumes. However, uncertainties remain over the problems of stress concentrations around the embedded shaft, and the behavior of the material under large impact loads, and after ageing.

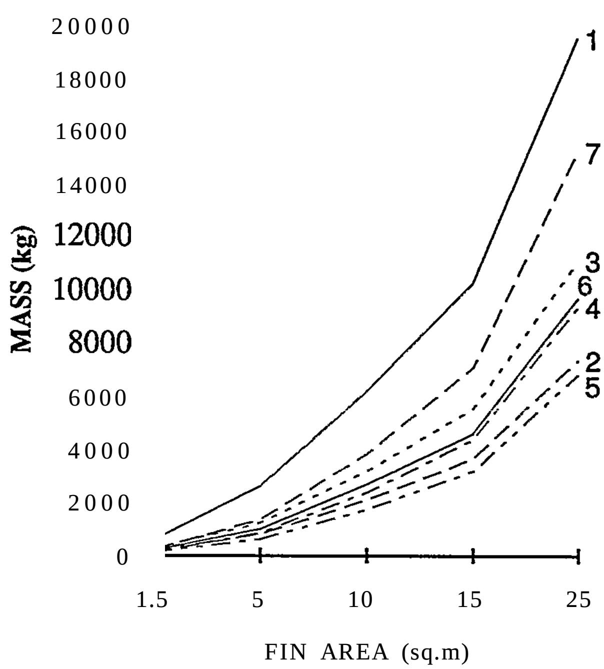


Figure 12. Mass of Fin Concepts

			CONCEPT NUMBER				
	1	2	3		5	6	7
EVALUATION CRITERIA				RANKINGS			
Accuracy of surface	7	5	3	4	5	1	1
Appearance of surface	5	6	3	4			
Resistance to erosion		5	1		2	1	1
Resistance to fouling	7	5		1	5	3	3
Resistance to seawater		6	4	3	5	1	1
Skin impact resistance	7	6		2	6	4	4
General resistance to impact	1	5	2		6		3
Static loads	1	5	4	7		3	2
Fatigue loads	1		4	7	3	3	2
Shock loads		5			6	3	2
Life	1		4	5	6	2	2
Shaft joint integrity	1	5	2	6	7	4	5
Overall design integrity	1	3	2	6		5	4
Materials track record	1	1	2	2	3	3	1
Manufacturing methods	1	2	3	3	3	4	4
Mass (ex. shaft) (kg)	581	150	240	141	107	186	224
(lb)	1278	330	528	310	235	409	493
Hydrostatic depth (m)	>5000	50	100	3500	150	50	3000
(ft)	16400	164	328	11484	492	164	9843
cost - fin and shaft	£7627	£4530	£4410	£5865	£6445	£ 2850	£ 3850
	$11974	$7112	$6924	$9208	$10119	$4475	$6045
- *fin* only	£ 5857	£ 3250	£3130	£4195	£ 5215	£ 1500	£ 2500
	$9195	$5103	$4914	$6586	$8188	$2355	$3925
Fin material cost	£ 732	£450	£ 400	£2175	£ 1535	£ 0	
	$1149	$707	$628	$3415	$2410	$ 0	$ 0
Fin sub-contract cost	£225	£0	£500	£ 500	£ o	£ 1500	£ 1800
	$353	$0	$785	$785	$0	$ 2355	$2826
Fin labor hours	178	100	74	48	128	0	20

Table II. Concept Evaluation

NO. 7- SOLID CASTING

Basic Form of Construction.

The requirements for a mold are as for Concept No. 6, but the material is cast solid without cores, and without the shaft in place. The casting is then bored to accept the shaft which can be adhesively bonded and retained axially. This method is included as an alternative to Concept No. 6 because it is currently used for the production of some small stabilizer fins. The advantages and disadvantages are much the same bug because cores are not included, and because the shaft has to be fitted, the mass and the cost are higher.

EVALUATION OF DESIGN CONCEPTS

The concept design proposals are evaluated against each of the evaluation criteria on the basis of the standard 1.5 m^2 (16 ft^2) roll stabilizer fin. The level of design detail available for the evaluation is of necessity quite low due to the number of concepts considered, and an element of subjective judgement is involved. It is therefore important not to place too much reliance on absolute values, but to use the figures mainly for comparisons. The results are summarized in Table II. The estimates of mass for each concept are plotted against fin area in Figure 12. The cost for each of the seven concept designs, broken down into labor and material cost for the fin itself plus a total cmt for the shaft is plotted in Figures 13 to 17 for the five different fin sizes considered.

Performance.

The performance requirements relate to the mass of the fin, the accuracy of the surface, and its resistance to the environment. The mass is lowest for foam structures without any significant steel elements, but their structural integrity is not well established, so

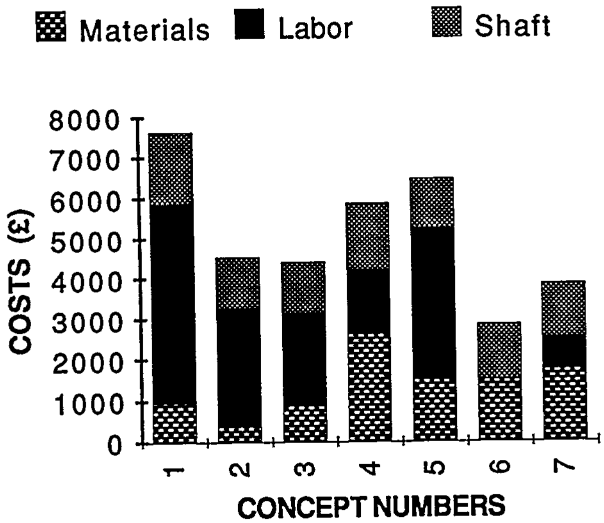


Figure 13. Costs for 1.5 sq.m Fin

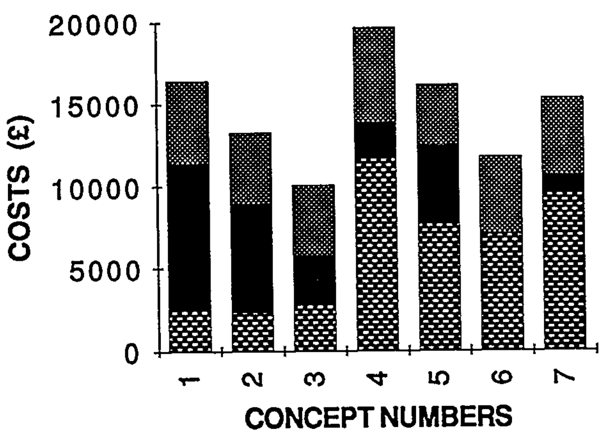


Figure 14. Costs for 5 sq.m Fin

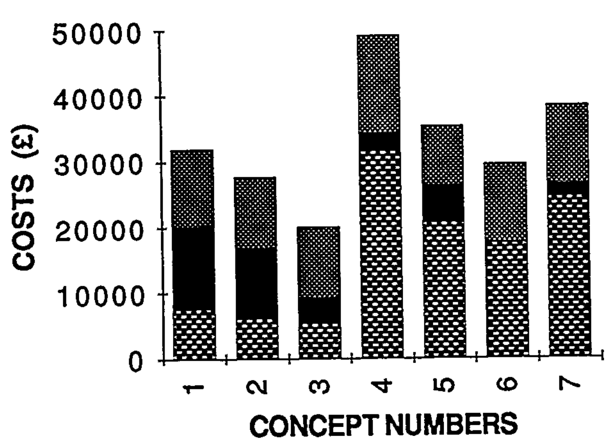


Figure 15. Costs for 10 sq.m Fin

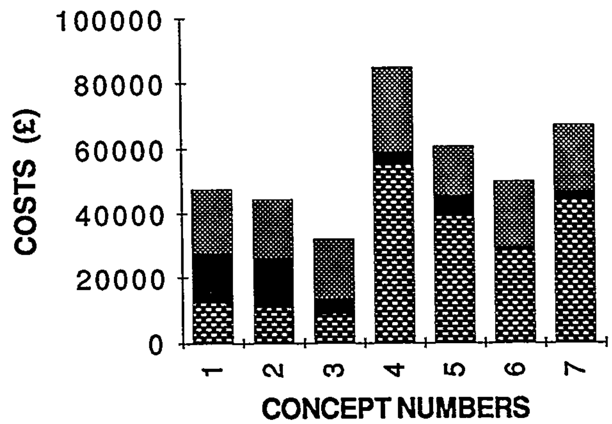


Figure 16. Costs for 15 sq.m Fin

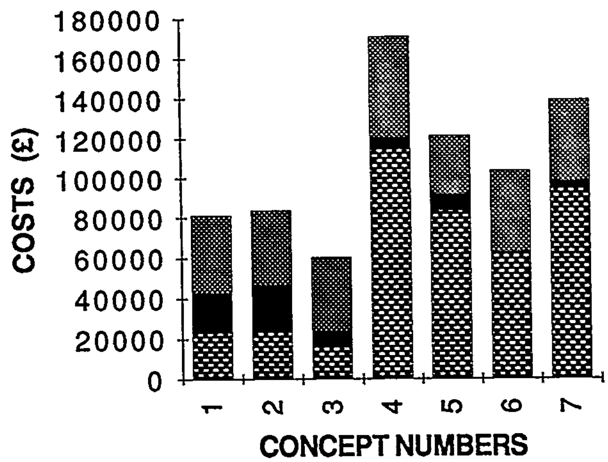


Figure 17. Costs for 25 sq.m Fin

Concepts 3 and 4 are preferred, even though their mass is higher. The cast nylon designs have low mass at small sizes, but this is not sustained as size increases. The best surface accuracy is achieved by the use of the full molds as in Concepts 6 and 7, with the mold plates of Concept 3 coming next. The sprayed polyurethane elastomer coating provides the best resistance to the environment. This can, of course, be applied as a surface finish to any of the concept designs, but with a low modulus foam substrate the risk of delamination of the coating under impact and cavitation loading is much reduced (5).

Strength

All the concepts are designed to carry the required loads, and to survive the specified life. The scantlings of the traditional steel construction are determined mainly by the requirement to avoid distortion, and to provide a corrosion allowance. The strength requirements

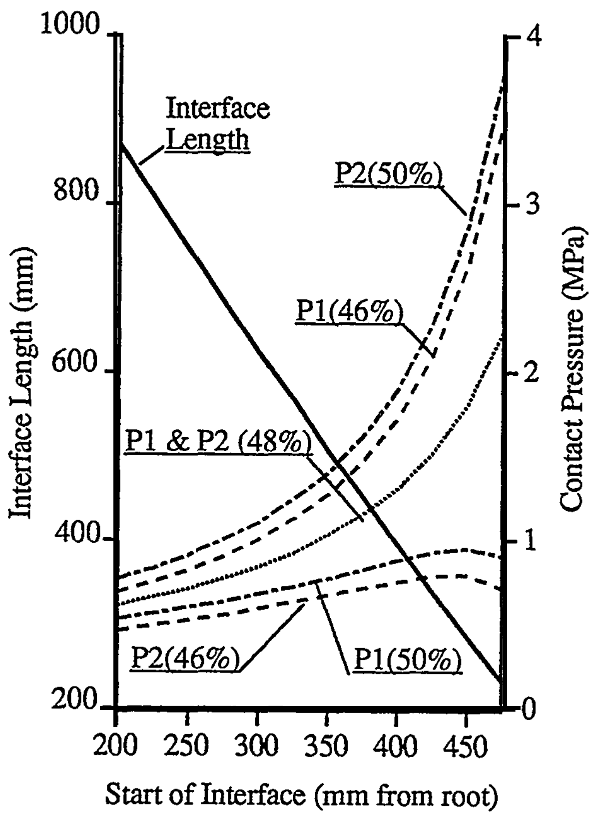


Figure 18. Contact Pressure at Shaft/Fin Interface

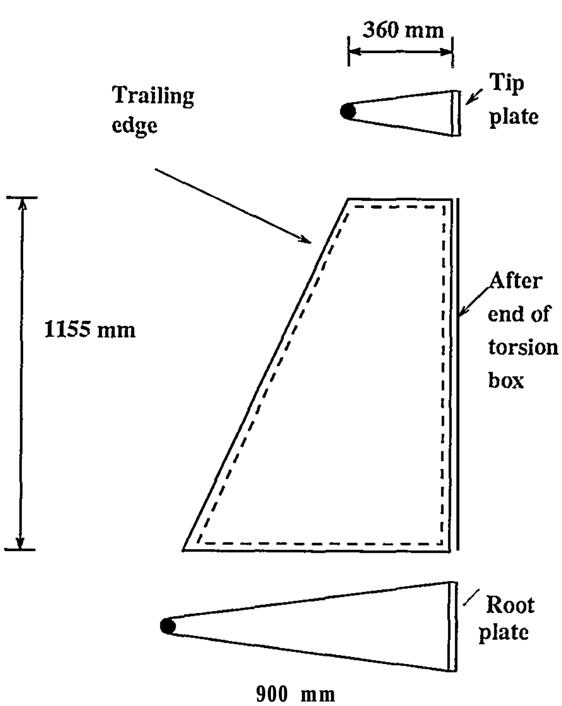


Figure 19. Sandwich Panel for 3D Analysis

foam and in the steel structure. A sandwich panel (see Figure 19) representing the foam-filled structure between the aft side of the torsion box and the trailing edge of a 1.5 m^z(16 ft^2) fin is analyzed with a number of different foam densities. The face plates of the sandwich are 3 mm (0.12 in) mild steel, while the root, tip and torsion box plates at the boundaries of the panel are all 10 mm (0.4 in) mild steel. A steel tube (outer diameter 35 mm (1.4 in), wall thickness 5 mm (0.2 in)) is welded between the tip and root plates, and ties tie top and bottom skin plates together at the trailing edge. The maximum thickness of the sandwich is 190 mm (7.6 in) at the root end closest to the torsion box, and the minimum thickness is 35 mm (1.4 in) at the trailing edge. The boundary condition for the analysis is that the face plates are fixed at the torsion box.

The load applied to the structure has a non-uniform pressure distribution. This is based on a triangular distribution of suction pressure on the upper side of the model, and a uniform distribution of pressure on the lower side, giving an equivalent total vertical reaction force of 66 kN (6.62 tonf) if applied to the entire fin surface. This corresponds to the maximum working load of the fin without safety factor, Both the foam and the steel are assumed to be isotropic, and to have linear elastic properties.

Calculations are carried out with foam densities of 100, 150 and 200 k~m^3(6.2, 9.4, 12.5 lb/ft^3). The foam used is a rigid closed cell polyurethane foam for which elastic modulus and density may be related to the properties of solid polyurethane by the equation given by Gibson and Ashby (6).

$$\frac{E_f}{E_s} = \varphi^2\left(\frac{\rho_f}{\rho_s}\right)^2 + (1-\varphi)\left(\frac{\rho_f}{\rho_s}\right) \qquad (4)$$

where Ef is the elastic modulus of the foam, Es is the elastic modulus for the solid material (1600 MPa (232000 psi)), ρ_f is the foam density, and ρ_s is the density of the solid material (1200 k~m , (74.88 lb/ft^3)). (p is the fraction of the volume of the polymer material that is contained within the cdl edges. Values given for this vary from 0.6 to 1.0, the latter value being used for this analysis. Poisson's ratio for the foam is assumed to be 0.25.

The analyses show a stress concentration in the steel face plates at the attachment points to the torsion box. The reason for this is that, at distances greater than about 100 mm (4 in) from the attachment points, the sandwich panel acts as a composite structure, with the foam core carrying the shear forces, and the face plates carrying the direct forces. Closer to the attachment points the face plates are held at a fixed distance apart by the stiff structure of the torsion box. As only small deformations can occur in the core material at this poing the foam does not carry much shear force, and the face plates behave largely as two separate plates with a consequent increase in stress. However, the maximum stress at the attachment point still varies with the foam density as shown in Figure 20. The

are easily satisfied, with the only disadvantage being that the rigidity of the design will tend to transmit impact forces direct to the shaft rather than absorb impact energy through distortion of the fin structure. The strength requirements can be easily met with the nylon castings, and in Concept 3 the steel frame and inner skin can be sized to carry the main loads so that the design is more tolerant of defects in the foam than those in which the foam is the principal structural element. There is some concern about fracture under impact loads of the solid nylon or foam designs when there is no GRP or steel outer skin.

The steel fabricated fin can withstand high hydrostatic pressures if it is flooded, while the nylon and syntactic foam have sufficiently high compressive strength to withstand high pressures as solid bodies. Syntactic foam could also be used with Concepts 3 and 5 to give a good hydrostatic depth capability.

Manufacture

Marine engineering companies and ship repair yards are mainly equipped to handle metal products, so the introduction of non-metallic composites would require investment in new facilities and in training, or the sub-contract of significant parts of the manufacture. Polyurethane foams, elastomers and GRP are all currently used in marine applications, and companies exist that can handle this sub-contract work. However, the technology is not complex, and the plant required is not excessively expensive, so it is a realistic proposition for a company to re-equip for a new method of manufacture. The steel fabrication contained in Concept 3 allows a good element of in-house work while still sub-contracting the polyurethane work.

Cost

Figures 13 to 17 show the shift in the balance of the costs as the fin area changes. The advantage of the solid cast foam or nylon fins at the smaller sizes is soon lost as the material cost outways the labor cost when size increases, since the mass of material varies with the cube of scale. Concept 3 establishes and maintains a consistent cost advantage at 5 m^2, (54 ft^2) size and above.

SELECTION OF PREFERRED DESIGN

In terms of strength, the steel fabrication holds a clear advantage, but the evaluation shows that savings can be made in cost and mass, and that improvements in performance can be realized. The cored cast nylon Concept 6 shows clear advantages at the smallest size, and cast nylon fins are already being produced at this size. However, over the full range of sizes, the best potential is offered by Concept 3, which has low cost, offers good reductions in mass, and provides the benefits of the polyurethane coating. The integral steel core fabrication allows this concept to be viewed as less radical than some, in that it provides reassurance in terms of strength capability, and in terms of familiar manufacturing processes.

DEVELOPMENT OF THE SELECTED DESIGN

Further work has been carried out on Concept No. 3 to establish its feasibility, and to obtain more accurate estimates of mass and cost. Areas of the design that have been studied in more detail follow.

Shaft Interface

The options to consider are a shaft with either circular sections bonded, or shrunk fit, into a tapered, or stepped, socket, or with tapered square sections bonded to a fabricated steel, torsion box. The latter option is attractive since the costs of casting and boring a circular socket are avoided, but difficulties with fabricating a square socket to the tolerances required for a bonded joint leads to serious consideration of the circular shaft. However, successful weld tests show that the required tolerances can be achieved and the square section fabricated torsion box has been adopted for design development.

The torsion box is designed with a three degree taper to match that of the surface of the fin, and the square sections of the shaft are machined to this same taper. The spanwise location of the interface between the torsion box and the shaft is selected so as to minimize the bending moments in the torsion box, and also to ensure that the external loads on the fin can be transferred to the shaft via a uniform contact pressure over the interface area, i.e. the pressure at one end, p_1, is equal to the pressure at the other end, p_2. The design is based on an assumed spanwise center of pressure of 48% of the span. However, this position is not accurately known, and may vary between 46% and 50% of the span, with a resulting change in the pressure distribution over the length of the shaft interface. Figure 18 shows the contact pressure at each end of the shaft interface plotted against the spanwise location of the start of the shaft interface; three possible positions of center of pressure are presented. The left-hand scale shows the required length of interface to ensure that $p_1 = p_2$ when the spanwise center of pressure is 48%. A start position of 400 mm (16 in) from the fin root is chosen to ensure a small variation of maximum contact pressure with changes in spanwise center of pressure. The required length of interface is then 394 mm (15.8 in).

Composite Beam Design

A three-dimensional finite element analysis is performed to enable the foam density to be selected on the basis of its influence on the stresses in the

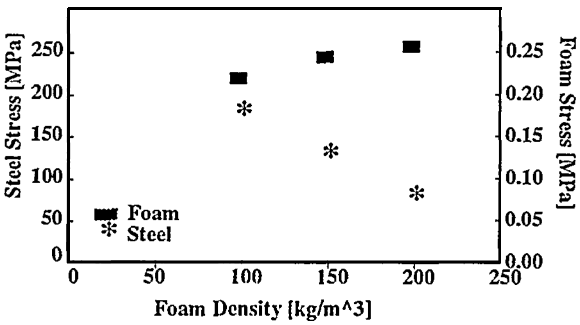


Figure 20. Maximum Stresses in Steel and Foam

figure also shows the maximum shear stress in the foam. This rises with increasing density because as the foam becomes stiffer, it carries more of the load.

On the basis of these results, a foam density of around 200 kg/m^3 (12.5 lb/ft^3) is selected, although further design work as described in the next section, is required in order to reduce the stress levels in the steel. The maximum shear stress of 0.26 MPa (37.7 psi) results in a good factor of safety against the theoretical shear strength of about 1.7 MPa (247 psi) derived from Gibson and Ashby, although tests on actual samples of foam with this density gave a shear strength of only 1.1 MPa (160 psi).

Torsion Box to Skin Plate Transition

The cantilever sandwich of steel/foam/steel is very efficient in carrying loads, but difficulties arise as the face plates approach the torsion box. Here they appear to act as two separate unsupported cantilever beams with a very rapid increase in stress. To obtain good fatigue performance it is important that stress in the welds is kept low. The main requirement is to establish additional stiffness in the transition region which extends some distance beyond the torsion box before the foam core fully supports the sandwich structure.

A number of different design solutions for the local steel structure are analyzed using the finite element method. A section at mid-span of the fin stabilizer is chosen for analysis. The displacement and the twist angle for the section are determined using three-dimensional analysis, and then used as boundary conditions for the four comer nodes of the torsion box. The load applied to the model is the non-uniform pressure distribution described earlier, but in this case the loading is equivalent to a total lift on the full fin of 137 KN (13.75 tonf). This represents a factor of two on the maximum design working load. The torsion box material, and that of the face plates of the sandwich structure, is mild steel, while the core material is polyurethane foam with a density of 210 kg/m^3 (13.1 lb/ft^3).

Two approaches to increasing the stiffness in the transition region are examined. First, the top and bottom plates of the torsion box are extended forward and aft, tapering over various lengths from a thickness of 10 mm (0.4 in) down to the 3 mm (0.12 in) required for the butt weld to the face plate. The second approach is to further stiffen the transition region by adding open triangular steel sections to the front and back of the torsion box, which extend into the foam, thus reducing the depth of foam in the transition region and increasing local stiffness.

The results of the analysis of these arrangements are presented in Figure 21. The plots show the direct stress acting in the chordwise direction on the top and bottom surfaces of the top and bottom face plates. The stress reversal across the thickness of the face plates in the region close to the torsion box shows that they are acting as independent beams, rather than a composite structure. The arrangement in Case c) gives the lowest stresses in way of the weld, and is adopted for the final design.

Analysis of Total Structure

With the required foam density found and tested, and the final design of the shaft and torsion box decided, a three-dimensional finite element analysis of the full inner structure of the 1.5 m^2 (16 ft^2 fin is carried out.

The model comprises 10mm (0.4 in) mild steel torsion box, root and tip plates, 3 mm (0.12 in) mild steel face plates, and 200 kg/m^3 (12.5 lb/ft^3) density foam core. The shaft interface extends from 400 mm (16 in) to 794 mm (31.8 in) from the root, and the model is fixed at the shaft where it passes through the root plate. The load applied to the structure is a suction load on the upper surface and a pressure load on the lower surface, with an approximately eliptical spanwise distribution, giving a center of pressure at 48% of the span. The total normal load in this analyses is 66 kN (6.63 tonf), representing the maximum normal working load.

The primary requirement of the analysis is to check the stress range in the various welds when the load changes between pressure and suction, as a result of a reversal in the angle of incidence of the fin to the water flow. The stress ranges are calculated by taking the difference between the stresses at corresponding nodes in the top and bottom plates. Since the top plate is loaded in suction and the bottom plate in pressure, the difference between them gives the design stress range under fatigue loading. Figures 22 and 23 show the stress range in the welds between the sandwich face plates and the forward and aft extensions of the torsion box plates. The stresses are given as direct stresses in the chord wise direction on the inner sides of the face plates, where the stress ranges are at their highest, The maximum stress range in a weld is approximately 65 MPa (9425 psi), which satisfies the requirement for a butt weld to achieve the target fatigue life of 10^7 cycles.

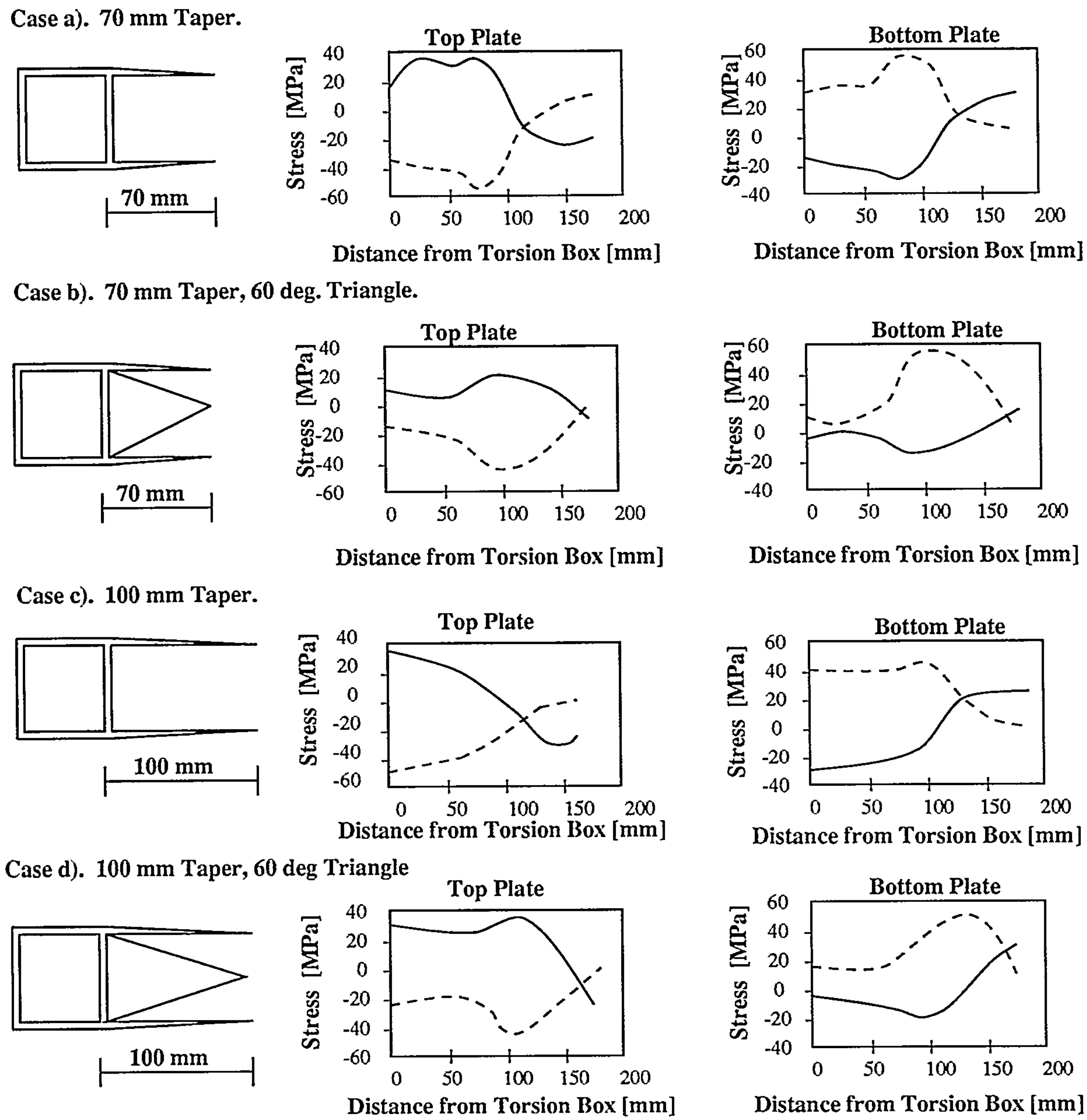


Figure 21. Stresses at Face Plate/Torsion Box Interface

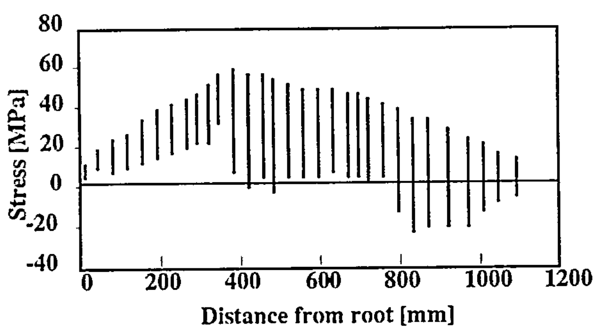


Figure 22. Stress Range in Weld for'd of Torsion Box

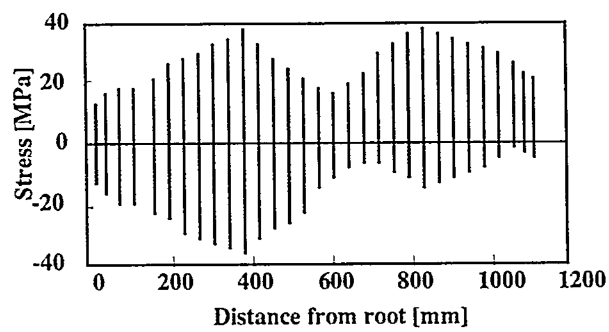


Figure 23. Stress Range in Weld aft of Torsion Box

Estimate of Mass

The detailed estimates give a total mass of 250 kg (550 lb) for the 1.5 m^2 (16 ft^2) fin, comprising 203 kg (447 lb) of steel work, 35 kg (77 lb) of foam, and 12 kg (26 lb) in the elastomer coating. This gives a 56% reduction in mass relative to the fabricated design, against a target reduction of 25%.

Estimate of Cost

The detailed estimates give a total cost for the 1.5 m^2 (16 ft^2) fin of £3900 ($6123), made up of £555 ($871) in material costs, and £3345 ($5252) in labor costs with overheads. This represents an estimated saving of 34% of the cost of the fabricated fin, against a target saving of 40%.

CONCLUSION

The conceptual design study identifies the requirements for hydrodynamic control surfaces, and shows that these can be met by a number of designs based on the use of composite materials as an alternative to the traditional steel fabrication. The concept designs are evaluated against a set of pre-defined criteria, and Concept No. 3 is selected as the design that provides the best alternative to steel fabrication. This design is developed in more detail in the form of a ship roll stabilizer fin with an area of 1.5 m^2 (16 ft^2). It offers the potential to reduce the cost of the fin itself (excluding the shaft) by 34%, and the mass by 56%, while also providing a more accurate and smooth surface that has better resistance to erosion, corrosion and marine fouling. Further work has been completed on the selected design to produce a detailed design procedure, and to build and successfully test the 1.5 m^2 (16 ft^2) prototype stabilizer fin. This work will be reported in future papers.

ACKNOWLEDGEMENTS

The work described in this report was carried out at the University of Glasgow in Scotland with funding from The Marine Technology Directorate, and support from Brown Brothers & Co. Ltd. of Edinburgh.

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